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None

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F1B

(54) **An air-compression, four-stroke internal combustion engine with direct fuel injection, turbo-charging and load-dependent exhaust gas recirculation**

(57) A self-ignition or spark-ignition engine has the inlet valve (3) opened during the exhaust stroke, the opening taking place not earlier than 30 degrees crank angle after bottom dead centre with the highest lift being between 15 and 30% of the maximum inlet valve lift. When the exhaust valve closes before top dead centre only a minimum opening of the inlet valve exists, e.g. 10% of the maximum lift, which is maintained until the start of normal inlet valve opening on the induction stroke. The additional inlet valve opening may be effected by an additional cam lobe or the inlet and exhaust valves may be electronically controlled using solenoids acting directly or on hydraulic or pneumatic actuators. The additional inlet valve opening provides for exhaust gas return at low loads and scavenging at high loads.

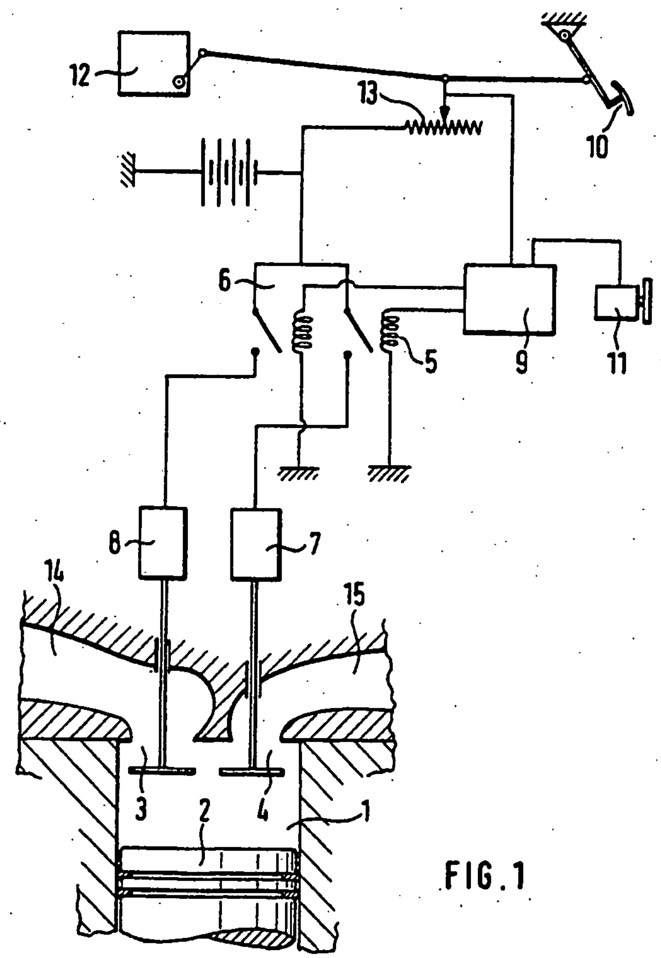

FIG. 1

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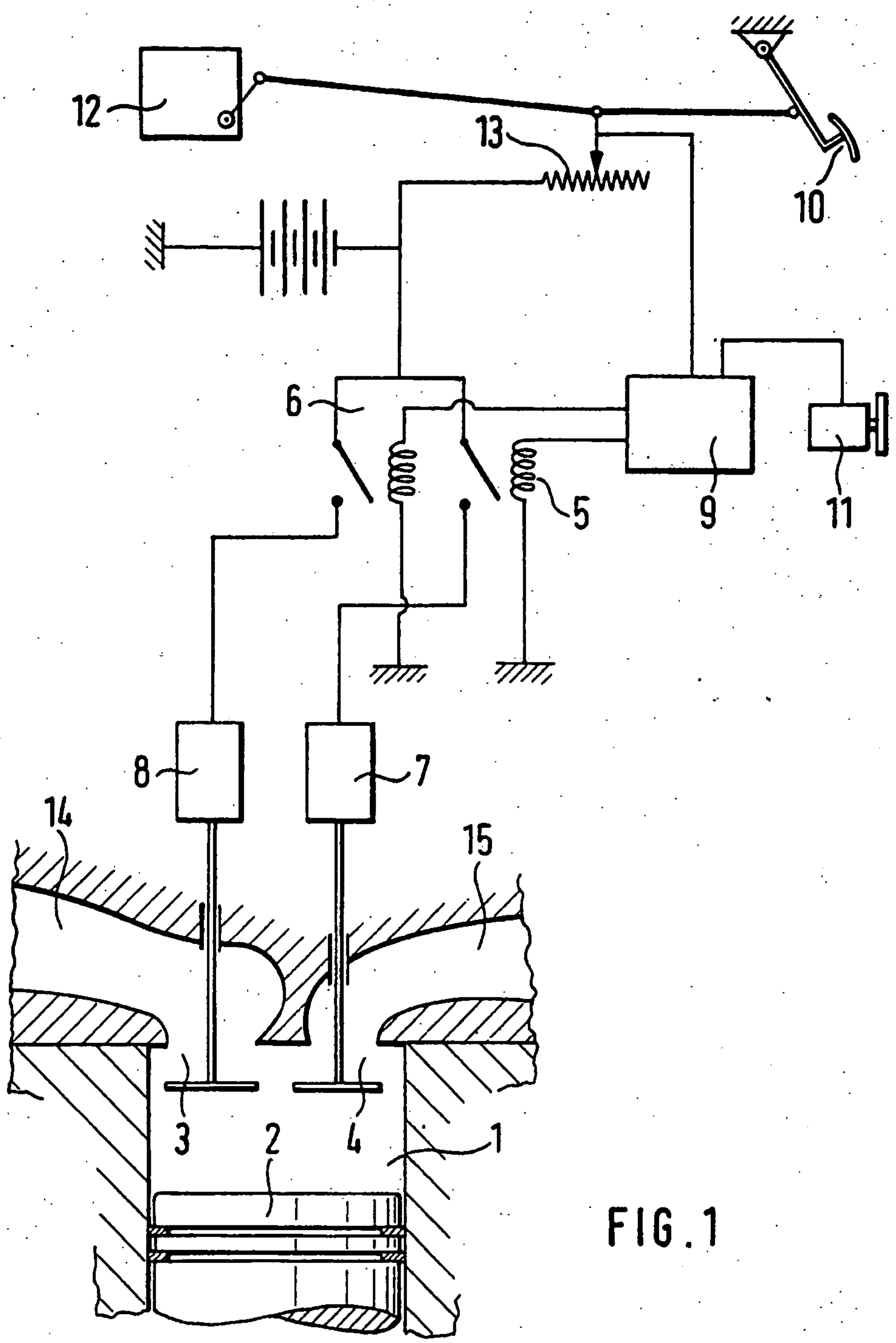

FIG.1

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FIG. 2

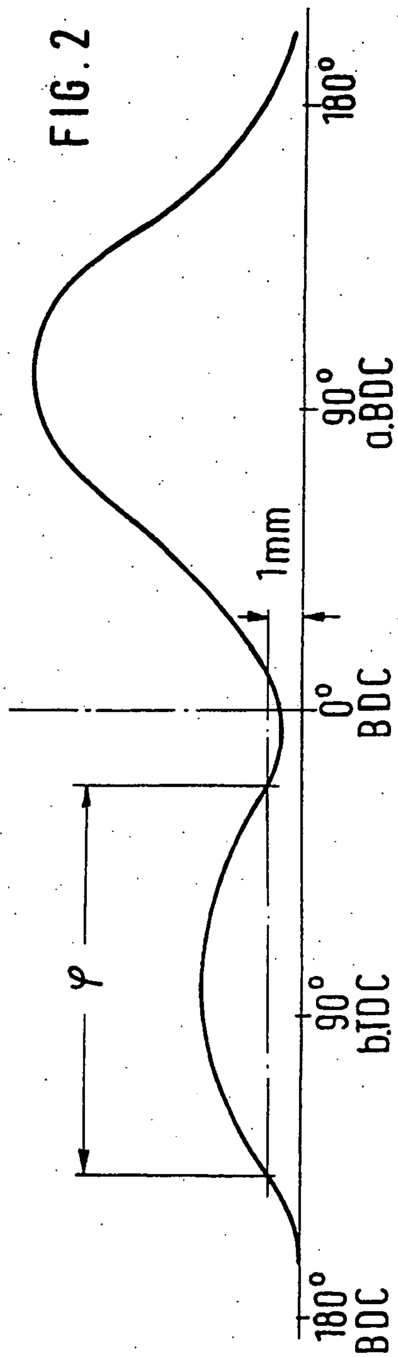


FIG. 4

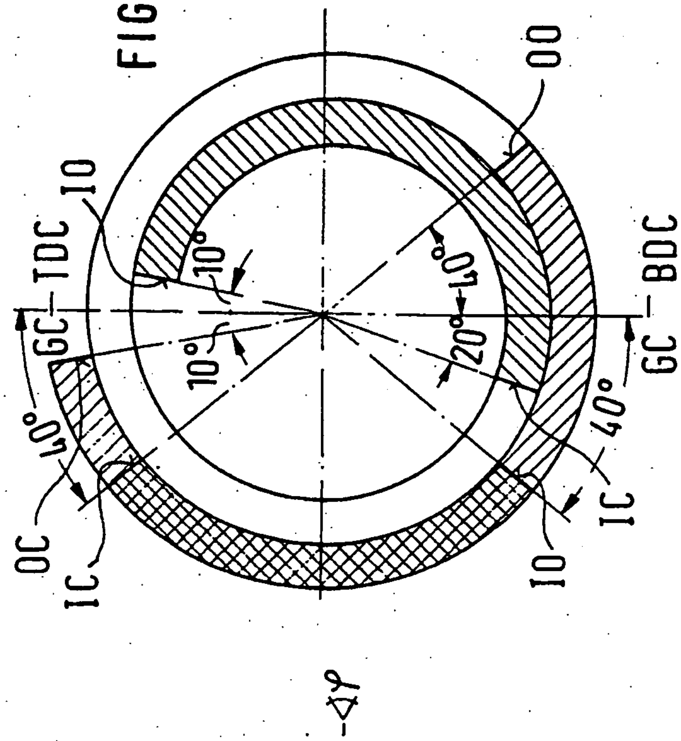
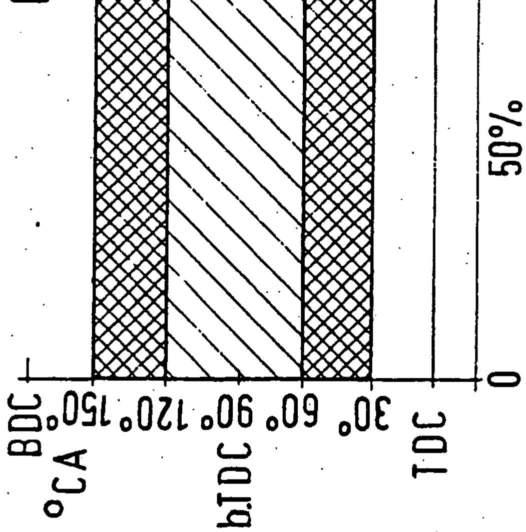


FIG. 3



SPECIFICATION

An air-compression four-stroke internal combustion engine with direct fuel injection turbo-charging and load-dependent internal exhaust gas recirculation

This invention relates to an air-compression self-ignition or spark-ignition four-stroke internal combustion engine with direct fuel injection, turbo-charging and load-dependent internal exhaust gas recirculation, effected at least in certain operating ranges by intervention in the gas-change system, wherein mixture formation is essentially controlled through the high-speed rotary motion of the fresh air charge produced in an inlet port and maintained in a combustion chamber in the shape of a solid of revolution.

Diesel engines have qualitative governing. There is no throttling and, consequently, volume flow is high. Owing to the difficulty of achieving mixture formation immediately before combustion in a minimum of time, provision is made for more air to be drawn into the combustion chambers of the engine than is actually necessary for combustion of the fuel injected into the combustion chambers. Generally, the proportion of inducted air is greater the lower the load of the engine. Moreover, combustion at low loads is sluggish and takes place at a low temperature level. Therefore, this is the range where the exhaust gases have undesired amounts of unburnt substances; above all, the great amount of oxygen is conducive to the formation of nitric oxides. This applies in particular to direct-injection engines.

In recent years, it has become general practice in turbo-charged internal combustion engines to return a proportion of the exhaust gases to the inlet in order to reduce emission of pollutants (nitric oxides and hydro-carbons).

Exhaust gas recirculation reduces the oxygen content of the air for combustion and, consequently, the effective excess air in the fresh gases. In other words, the reaction kinetics of combustion are interfered with through the oxygen concentration of the cylinder charge whereby the combustion process and the exhaust gas composition are influenced.

Another very important apsect of exhaust gas recirculation is the reduction of the ignition lag which means the time from the start of injection of the fuel to the start of combustion. Ignition lag is a consequence of the higher final compression temperature resulting from the higher fresh gas inlet temperature. Apart from other advantages, e.g. the reduction of ignition noise, the shortening of the ignition lag results in an improvement in combustion which, in turn, decreases the emission of pollutants.

In an exhaust gas recirculation control system for Diesel engines, it is desirable for the amount of air replaced by exhaust gas to be proportional to the amount of surplus air relative to actual air requirements for the combustion of the injected fuel, such that a maximum amount of surplus air is removed from the air flow supplying the engine cylinder without causing unsteady combustion of the fuel in the cylinders: the object being to achieve maximum efficiency throughout the operating range of the engine with respect to control of nitric oxide emission.

The maximum rate of exhaust gas recirculation is required in at lower part loads because this is where the greatest amount of excess air exists. At full loads, however, a high rate of exhaust gas circulation would reduce the output which the engine can attain because there is only a small amount of excess air. For this reason, it is necessary for the rate of exhaust gas recirculation to be controlled so that the proportion of recirculated exhaust gas decreases as the load of the engine increases, and no exhaust gas at all is recirculated at full load.

A distinction is made between external and external exhaust gas recirculation. In the case of external recirculation, the exhaust gases are returned from the exhaust gas ports *via* pipes and control devices into the inlet port. In contrast to this, internal exhaust gas recirculation can be implemented in a simpler manner by appropriate intervention in the gas change system. Therefore, internal recirculation offers certain advantages over the external concept; especially, it affords advantages with respect to hydro-carbon emission at low loads (due to the hotter exhaust gas). Last but not least, it will also assist cold starting and warming-up of the engine.

Intervention in the gas change system in the case of internal exhaust gas recirculation can be effected in various ways (see German AS 12 22 735 and German PS 12 42 044), the means employed including, inter alia, double-rise cams for controlling the gas change valves (see for instance, German AS 14 01 228, German OS 21 25 368, German OS 26 38 651, German OS 27 10 189, German PS 17 51 473).

Of the specifications mentioned, only the German Patent 12 42 044 covers a turbo-charged internal combustion engine.

The present invention relates to this type of internal combustion engine, using internal exhaust gas recirculation in line with the generic term defined initially.

An object of the invention is to improve such an internal combustion engine in a manner that, on the one hand, exhaust gas recirculation does not cause any weakening of the air swirl energy (charge swirl energy) at low loads which would spoil the desired exhaust gas improvements and that, on the other hand, assisted cylinder scavenging would be obtained at higher loads.

This object is achieved in that the inlet valve is briefly opened during the exhaust stroke, the opening being timed not earlier than 30 degrees crank angle after bottom dead centre of the gas change cylinder and the highest lift being between 15 and 30% of the maximum inlet valve lift and in that at the latest when the exhaust valve closes (briefly before gas change BDC) only a minimum opening of the inlet valve exists which is maintained until the start of the suction stroke (briefly after gas change TDC).

Due to the brief advance opening of the inlet valve during the exhaust stroke, several improvements are obtained. At low loads, advantageous internal exhaust gas recirculation is obtained which, at the upper loads is automatically reduced and then removed due to the pressure conditions of charge air and exhaust gas. As a rele, the charging pressure up to about 30% engine load is lower than the cylinder pressure: at higher engine loads up to full load, however, the charging pressure is higher. This means that at these loads, air will then flow through the cylinder into the exhaust system. This provides assisted scavenging of the combustion chamber (during the exhaust stroke only) and additional cooling of the piston and the cylinder head. It is possible to do without scavenging at top dead centre which is usual in turbo-charged engines in order to ensure adequate scavenging of the combustion space and, consequently, to construct the piston without valve recesses. The absence of valve recesses offers advantages in maintaining the rotary air motion in the combustion space and benefits in shaping the swirl port. Without valve recesses, it is also possible to allow a higher thermal loading of the piston because there is no stress concentration effect due to the valve recesses.

The exhaust gas which (at low load) is pushed back into the suction ports is drawn in again during the induction stroke. Since internal exhaust gas recirculation is effected by opening the inlet valve during the exhaust stroke and not by re-opening the exhaust valve which would also be possible during the induction stroke, there is no interference with (reduction of) the air swirl energy. In other words, the gas mass which flows through the swirl port of a direct-injection internal combustion engine and determines the air swirl energy in the cylinder (combustion chamber), is not decreased as the rate of exhaust gas recirculation is increased. Since no exhaust gas recirculation is desired at high engine loads, but only the scavenging of the combustion space, advance opening of the inlet valve will start only at about 30 to 40 degrees crank angle after gas change BDC when the cylinder pressure has already dropped to atmospheric. In order to prevent too much exhaust gas being recirculated at low loads (and to prevent too much fresh air flowing through the cylinder at higher loads), the highest lift of the inlet valve during the exhaust stroke is limited to 15 to 30% of the maximum inlet valve lift. This lift is completed at the latest when the exhaust valve closes (briefly before gas change TDC), the inlet valve subsequently remaining open minimally until the start of the suction stroke briefly after gas change TDC (not more than 1/10 of the maximum inlet valve lift). This affords advantages in so far as there are no interruptions in the gas mass flow.

Control of the inlet valve during the exhaust stroke can be effected advantageously either fully electronically or a pre-lift nose which is known per se (on the inlet valve cam).

In a manner known per se, the air velocity of the rotating air for combustion—referred to the pitch diameter (0.7 times cylinder or piston diameter) and maximum inlet valve lift as well as 10 m/sec. mean piston speed—has a tangential component of 30 to 50 m/sec.

An embodiment of the invention will now be described with reference to the accompanying drawings, wherein:

Figure 1 is a schematic part-sectional view of a cylinder and control system for an internal combustion engine,

Figure 2 is a graph of valve lift (opening function) of the inlet valve over a full engine cycle,

Figure 3 is a graph showing the inlet valve opening angle during the exhaust stroke, and

Figure 4 is an example of a typical valve timing (opening times of the valves versus the crank shaft angle).

Figure 1 shows schematically the areqa around the top of a cylinder of an air-compression, self-ignition or spark-ignition four-stroke internal combustion engine with direct fuel injection and turbo-charging. A piston 2 (with combustion chamber) is movable in a cylinder bore 1. Valves (inlet valve 3 and exhaust valve 4) control the engine and are actuated by variable valve control means in the form of electro-mechanical relay systems comprising relays 5, 6 and solenoids 7, 8 serving as final control members.

Engine characteristics for various operating modes, i.e. the timings for the inlet valves 3 and exhaust valves 4 as well as fuel injection and ignition, are stored in a control unit 9. In response to a control input (accelerator pedal 10), speed information 11 (speed signal transmitter) and additional parameters, such as engine component temperatures, cooling water temperature, combustion air temperature, etc., these engine characteristics are fed to the valve actuators as well as the injectors and ignition devices. The control unit 9 determines the opening period of the valve 3, 4 in degrees crank angle as a function of the engine load which, as already mentioned, is determined by the driver actuating the accelerator pedal 10 controlling the rate of injection of

the injection pump 12, and transmitting a signal to the control unit 9 *via* the converter 13. A speed-dependent correction is transmitted, as already mentioned, by the speed signal transmitter 11 to the control unit 9 as a function of the engine speed.

According to the invention, the inlet valve 3 is briefly opened during the opening period of the exhaust valve 4 (i.e. during the exhaust stroke) dependent on the engine load. The smaller the engine load, the longer the opening period of the inlet valve 3 during the exhaust phase. The improvements obtained are as follows:

1. At low load:

An amount of exhaust gas which is dependent on the engine load is displaced into the induction port 14 during the exhaust phase (the exhaust gas back-pressure being higher than the charge air pressure). This mount of exhaust gas is drawn in again during the subsequent induction phase. The charge in the cylinder then consists of a mixture of air and exhaust gas and now contains only the amount of oxygen actually required at this engine load.

Since the total volume flow through the inlet port during the inlet phase is maintained, there is no weakening of the high swirl energy required in direct-injection engines for mixture formation. As a result of the lower volume flow through the exhaust port 15 at lower engine load (during the exhaust phase), the gas change work is smaller since pressure losses in the exhaust system are less. The amount of fresh air required is correspondingly smaller and consequently, the load on the air cleaner is less. Since the charge in the cylinder at the end of the induction stroke at low load consists of fresh air and exhaust gas, the effect of hot exhaust gas recirculation is obtained. Thanks to this, exhaust gas emissions at low load are reduced for two reasons:

a) The concentration of some pollutants in the combustion space, which is partially filled with hot exhaust gas, is less. Nitric oxides, in particular, are suppressed during combustion because the oxygen content of the charge is less. Due to the hotter charge in the cold engine and at low load, concentrations of unburnt matter (hydro-carbons) are also less.

b) Emissions by volume are less because the exhaust gas is partly retained in the engine.

c) Ignition lag is less.

2. At higher load (upwards of about 40% of rated load):

Due to the converse pressure conditions (the charge air pressure is now higher than the exhaust gas pressure), there is no backflow of the exhaust gases into the induction port, but instead fresh air is drawn into the cylinder. As a result, there is intensive scavenging of the cylinder during the exhaust stroke. This obviates the need for the usual scavenging of the cylinder during valve overlap (at gas change TDC), which has the drawback that valve recesses are required in the piston.

This is particularly advantageous for direct-injection engines, because the valve recesses hinder the maintenance of air swirl energy until the ignition top dead centre.

In order to prevent exhaust gas being recirculated at higher loads due to the high cylinder pressure, advance opening of the inlet valve 3 starts only at about 30 to 40 degrees crank angle after gas change BDC, because at this stage the cylinder pressure will have dropped to atmospheric pressure.

The highest lift of the inlet valve 3 during the exhaust stroke is limited to 15 to 30% of the maximum inlet valve lift. On the one hand, this prevents excessive exhaust gas recirculation (in the low load ranges) and, on the other hand, no excessive scavenging (in the higher load ranges). The advance opening of the inlet valve 3 is terminated at the latest when the exhaust valve closes (briefly before gas change TDC): the inlet valve 3 subsequently remains minimally opened until the start of the induction stroke (briefly after gas change TDC), a maximum of 1/10 of the maximum inlet valve lift). As a result, there are no interruptions of the flow.

Figure 2 shows the lift function of the inlet valve 3 over a complete engine cycle (referred to a given engine load or speed). Double opening of the inlet valve can be effected either fully electronically, (see Figure 1) or can be implemented in a simpler manner by providing an additional nose (pre-lift nose) on the inlet cam. The opening period, or more aptly, the opening angle (in phase and duration) is denoted by the symbol ϕ. It can also be seen that the opening angle is calculated only upwards of 1 mm valve lift.

Figure 3 deals in somewhat greater detail with the determination of the inlet valve opening angle in the case of supercharged engines. In this graph, the mean effective pressure in % (on the abscissa) is plotted against the crank angles CA (on the ordinate). In the upper and lower range of the graph there is a respective cross-hatched bank, the upper band relating to the opening range and the lower band to the closing range of the inlet valve 3. These bands indicate the differences which are intended for optimization of various turbocharger selections or engine speeds. In the example illustrated, the opening angle extends maximally from 150 degrees crank angle before TDC to 30 degrees crank angle before TDC and minimally from 120 degrees crank angle before TDC to 60 degrees crank angle before TDC. The cross-hatched band from 150 degrees crank angle before TDC to 120 degrees crank angle before TDC relates to the opening range and the cross-hatched band

from 60 degrees crank angle before TDC to 30 degrees crank angle before TDC to the closing range of the inlet valve 3.

An example of a typical valve timing (for an engine with a nominal speed of 3000 rpm) is given in Figure 4 from which can be seen the opening periods (here again the timings are given at 1 mm valve lift) of the inlet valve 3 and the exhaust valve 4 against the crank angle position.

During the exhaust phase, the inlet valve starts to open (IO) 40 degrees crank angle after GC-BDG or, to put it differently, 140 degrees crank angle before GC-TDC, FC-BDC meaning gas change bottom dead centre and GC-TDC gas change top dead centre. The opening time OO of the exhaust or outlet valve is 40 degrees. Crank angle before GC-BDC. the closing time OC at 10 degrees crank angle before GC-TDC. The inlet valve closes (IC) already again at 40 degrees crank angle before gas change TDC. Consequently, there is an overlap of 100 degrees crank angle of the opening period of the inlet valve with the opening period of the exhaust valve (210 degrees crank angle). During the induction stroke, the inlet valve opens (IO) normally at about 10 degrees crank angle after gas change TDC: the closing time (IC) is at 20 degrees crank angle after gas change BDC (total opening period of the inlet valve being 190 degrees crank angle). The exhaust valve remains closed throughout this period.

For engines in which hydraulic or pneumatic power is available, control of the opening motions of the valves 3 and 4 is by means of suitable electrically actuated valves or hydraulic or pneumatic final control elements.

Finally, it may be useful in the case of the present invention, in particular during cold starting and during the warming-up phase to provide an additional adjustable valve installed in the exhaust system and/or an additional adjustable valve installed in the induction system in order to increase the amount of exhaust gas supplied (increasing the exhaust back-pressure). Such throttling action in the exhaust/or induction system will as a byproduct provide the conventional pressure wave effect by which exhaust gas recirculation and/or scavenging can be intensified.

CLAIMS

1. An air-compression four-stroke internal combustion engine with direct fuel injection, turbo-charging and load-dependent internal exhaust gas recirculation effected at least in certain operating ranges by intervention in the gas change system where mixture formation is essentially controlled by the high-speed rotary motion of the fresh charge produced in an inlet port and maintained in a combustion chamber in the shape of a solid of revolution, wherein means are provided for briefly opening the inlet valve during the exhaust stroke, the opening taking place not earlier than 30 degrees crank angle after gas change bottom dead centre and the highest lift is between 15 and 30 % of the maximum inlet valve lift, and at the latest when the exhaust valve closes (briefly before gas-change top dead centre) only a minimum opening of the inlet valve exists which is maintained until the start of the suction stroke (briefly after gas-change top dead centre).

2. An internal combustion engine as claimed in Claim 1, wherein the inlet valve which up to the time the suction stroke starts is minimally opened is opened not more than 1/10 of the maximum inlet valve lift.

3. An internal combustion engine as claimed in Claims 1 and 2, wherein control of the inlet valve during the exhaust stroke is either fully electronic or effected by means of a pre-lift nose on the inlet cam.

4. An internal combustion engine as claimed in Claim 1, wherein the air velocity of the rotating air for combustion in the cylinder—referred to the pitch diameter (0.7 times cylinder or piston diameter)—with the inlet valve fully opened and a mean axial piston speed of 10 m/sec. has a tangential component of 30 to 50 m/sec.

5. An internal combustion engine substantially as herein described with reference to the accompanying drawings.

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